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VANE DYNAMICS ANALYSIS OF A TILTED VANE ROTARY COMPRESSOR

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ABSTRACT

An analytical model of vane dynamics in a tilted vane rotary compressor is presented. With the geometric relationship of the vane in a tilted slot, and the roller, the kinetic equations were derived for evaluating the reciprocating inertial force of the vane, and the chamber volume equations were obtained for calculating pressure history inside the compression chamber. The empirical friction coefficients were used to figure out the friction forces between the vane and slot and between the vane tip and roller. The model was compared with a dynamics program for regular rolling piston rotary compressors and displayed enough accuracy to predict vane dynamics for both regular and titled vane rotary compressors. A developed computer program was utilized to predict the effects of vane tilted angle on vane tip normal force and friction energy loss. The calculated results show that with a certain positive vane tilted angle, the vane tip normal force has the minimum value, but the vane friction loss has no significant change.

INTRODUCTION

Generally, in a rolling piston rotary compressor the centerline of the vane goes through the center of the cylinder bore. In this structure, when the roller pushes the vane into the vane slot, i.e. the crankangle being larger than 180 degrees, the force acted on the vane tip by the roller is not parallel to the vane movement direction, therefore forms a component force perpendicular to vane centerline. This component force increases the friction force between vane and the slot, and also causes vane deflection. In order to raise mechanical efficiency and reduce wear between vane, cylinder and roller, a rolling piston compressor with tilted vane arrangement was figured out. In this compressor, the vane centerline has an angle with the line connecting the bore center and the intersecting point of cylinder bore and vane centerline, shown in Fig. 1 a). In this way, it can be imaged that the friction force between vane and the slot and the possible vane deflection could be reduced. As a fundamental research for this compressor structure, the kinetic and dynamic characteristics, the tip load, and the friction power of the vane have been analyzed based on its new geometric relationship. This paper presents an analytical model of vane dynamics for this tilted vane rolling piston compressor. The model includes vane kinetic, dynamic, and related thermodynamic equations. A computer program was also developed to predict the effects of vane tilted angle on vane tip normal force and friction energy loss.

ANALYTICAL MODELS

Vane Movement

The geometric relationship of a tilted vane rotary compressor is shown in Fig. 1 a). According to the coordinate system used in the figure, the equation of roller outer circle is

$$(x-x_0)^2+(y-y_0)^2=r^2 \tag{1}$$

where r is roller outer radius, x_0 and y_0 are coordinates of roller center. The position of roller center can be expressed as $x_0=e\sin\theta$, $y_0=e\cos\theta$, where θ is crankangle and is a function of time, e is the eccentricity.

Assume that the thickness of vane can be neglected at this moment, then the vane equation is

$$y=R+x\tan(\pi/2-\gamma) \tag{2}$$

where R is cylinder bore radius, γ is vane tilted angle, shown in Fig. 1 a). The vane tilted angle is defined as the positive when the vane extending direction is forward the discharge port side.

From equation (1) and (2), the coordinates of the intersecting point of roller outer circle and vane can be obtained, i.e.

$$x=(-b+\sqrt{b^2-4ac})/2a, \qquad y=y_0+\sqrt{r^2-(x-x_0)^2} \tag{3}$$

where a=1+tan(π/2−γ), b=2[(R-y_0)tan(π/2−γ)-x_0], c=x_0^2-r^2+(R-y_0)^2.

Vane extension, which is defined as the distance from vane tip to the intersecting point of vane centerline and cylinder bore, shown in Fig. 1 a), can be written as

$$h_e=\sqrt{x^2+(y-R)^2} \tag{4}$$

Thus from the first and second derivatives of equation (4) with respect to time, vane velocity v_e and acceleration a_e can be derived as

$$v_e=[xx'+(y-R)y']/h_e \tag{5}$$

$$a_e=[(x')^2+(y')^2+xx''+(y-R)y''-v_e^2]/h_e \tag{6}$$

where x' and y' are the first derivatives of equation (3) with respect to time, and x'' and y'' the second derivatives.

Gas Force on Vane Side Surfaces

Same as a regular rolling piston compressor, the tilted vane divides the crescent shape volume, formed by roller and cylinder, into suction and compression chambers. The suction chamber volume is

$$V_s=L\{R^2[\theta-\varepsilon^2\alpha-\varepsilon(1-\varepsilon)\sin\alpha]+h_e\gamma\}/2 \tag{7}$$

where L is the cylinder height; ε is radius ratio of roller and bore (ε=r/R); the angle α is shown in Fig.1 a) and

$$\alpha=\theta+\sin^{-1}[(1-\varepsilon)\sin\theta/\varepsilon]$$

Neglect vane thickness, then the compression chamber volume will be

$$V_c=L\{R^2[(2\pi-\theta)+\varepsilon^2\alpha+\varepsilon(1-\varepsilon)\sin\alpha]-2\pi r^2-h_e\gamma\}/2 \tag{8}$$

For suction process, cylinder pressure is assumed to be equal to the suction pressure p_s. For compression process, cylinder pressure is calculated by the polytropic process equation, i.e.

$$p=p_s(V_{co}/V_c)^n$$

where V_{co} is the compression chamber volume when the compression process starts; n is the polytropic compression exponent. For discharge process, the cylinder pressure is assumed to be equal to the discharge pressure p_d. Therefore, the gas force acted on the vane side surfaces is

$$F_{vg}=h_eL(p-p_s) \tag{9}$$

Vane Reciprocating Inertial Force

Define that vane displacement forward the cylinder center is positive, then the vane reciprocating inertial force will be

$$F_{ve}=-m_va_e \tag{10}$$

where m_v is the equivalent vane mass.

Vane Reciprocating Friction Force

Assume the vane reciprocating friction force is proportional to the gas force acted on the vane side surfaces, then the vane reciprocating friction may be expressed as

$$F_{fr}=\pm C_f F_{vg} \tag{11}$$

where C_f is the friction coefficient; when the vane velocity is positive, the minus sign is used in equation (11), otherwise the plus sign is used.

Vane Spring Force

The vane spring force may expressed as

$$F_k=C_k(h_0+2e-h_e) \tag{12}$$

where C_k is the spring constant and h_o the pre-compressed length.

Gas Force in Vane Moving Direction

The vane back gas force, shown in Fig. 1 d), is

$$F_{db}=b_v L p_d \tag{13}$$

where b_v is vane thickness. In Fig. 1 b)

$$AB=e+(h_e-r_k)\sin\gamma/\sin\theta \tag{14}$$

where r_k is vane tip radius. From $AB/\sin\alpha_0=(r+r_k)/\sin\theta$, then

$$\alpha_0=\sin^{-1}\{[e\sin\theta+(h_e-r_k)\sin\gamma]/(r+r_k)\}, \qquad \beta=\alpha_0+\gamma.$$

Therefore, $b_{v2}=0.5b_v+r_k\sin\beta$, $b_{v1}=b_v-b_{v2}$, shown in Fig. 1 d). The gas force acted on the vane tip are $F_{dc}=b_{v2}Lp$, in compression chamber side; and $F_{ds}=b_{v1}Lp_s$, in suction chamber side. The total gas force acted in vane moving direction is

$$F_d=F_{db}-F_{dc}-F_{ds} \tag{15}$$

Vane Force Analysis

The composite force acted in vane moving direction is

$$F_v=F_d+F_{fr}+F_{ve}+F_k \tag{16}$$

From the geometric relationship shown in Fig.1 b) and c), following related forces can be figured out.

$F_n=F_v/\cos\beta$	(vane tip normal force)	(17)
$F_f=C_t F_n$	(vane tip friction force)	(18)
$F_t=\sqrt{F_n^2+F_f^2}$	(composite force of vane tip normal force and friction force)	(19)

VERIFICATION OF THE COMPUTER PROGRAM

Based on the above compressor models, a computer program was developed. This program can be used to predict vane extension, velocity and acceleration, cylinder pressure history, vane tip forces and vane friction power. When the program was set up, the roller rotation speed was assumed to be a fifteenth of that of the crankshaft, and the friction coefficient to be fifteen percent. In order to verify the accuracy of this program, it was compared with a standard program for rolling piston compressor. At this time, the vane tilted angle γ was set up as zero degree, that is, the tilted compressor was converted into a regular rolling piston compressor. The calculations were carried out for part of Tecumseh RG series compressors under the following conditions: refrigerant being R22, rpm 3450, evaporating temperature 32 °F (0 °C) and condensing temperature 150 °F (65.6 °C). Table 1 shows the calculated results of maximum vane tip loads. A comparison of the 'standard' program and 'tilted' program shows that so far as the maximum vane tip loads (normal forces) is concerned, two programs have very close results and their relative errors are less than 3%. This verifies that the program developed in this project may accurately predict vane dynamic characteristics of tilted vane rotary compressors.

CALCULATED RESULTS AND DISCUSSIONS

The calculations for a 5000 Btu model compressor under the conditions same as the above also give us an amount of important information. Figure 2 shows the changes of the vane extention with crankangles for regular compressor and that with a 15 degree vane tilted angle. The maxmum extension of the vane is about 0.47 mm larger for the tilted angle case than that for the regular case, and the crankangle reaching to the maxmum extension is about 10 degrees earlier than that for the regular case. Figure 3 displays the changes of the vane tip normal force. With a 15 degree vane tilted angle, the maximum vane tip loads are about 20 N reduced. This is of benefit to decrease vane deflection. Figure 4 shows the transient vane friction power. Even though the tilted vane lessens the maximum vane tip loads, due to the influence of vane velocity it does not reduce the transient friction power. Figure 5 shows the relation between the maximum vane tip normal force and vane angle for some models of compressors. It is found that when the vane tilted angle is close to 15 degree, the maximum vane tip normal force gets the minimum value. Therefore, as concerns maximum vane tip load, the vane having 10-15 degree angle will be favorable. However, for the predicted vane friction power, shown in Figure 6, the minimum vane friction power appears between 0 and 5 degree vane angle for the compressors. According to the predicted results, the tilted vane structure has no significant effect to reduce friction energy loss and improve the mechanical efficiency.

CONCLUSIONS

1. A FORTRAN computer program, which can do vane dynamic analysis for both regular and tilted rotary compressor, has been developed. The comparison between this program and a standard regular vane program shows that this program has enough accuracy to predict vane dynamic characteristics for both regular and tilted vane rotary compressors.
2. The calculated results show that when the vane angle is close to 15 degrees, the maximum vane tip normal force gets the minimum valve. Therefore, by using tilted vane structure, the durability and reliability of the vane and roller may be improved.
3. The predicted results show that the tilted vane structure has no significant effect to reduce friction energy loss and improve the mechanical efficiency.

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REFERENCE

Tatashi Yanagisawa, Takashi Shimizu and Tatsuya Horika, "A Study on Starting Characteristics of a Rolling Piston Type Rotary Compressor," Proceeding of the 1986 International Compressor Engineering Conference -

At Purdue, West Lafayette. July, 1986, pp. 808-823.

Table 1 Comparison on Calculated Maximum Vane Tip Loads

Capacity Btu/h(W)	Standard lbf(N)	Tilted lbf(N)	Error (%)
5100(1495)	41.0(182)	40.9(182)	0.2
5600(1641)	45.0(200)	45.0(200)	0.0
6000(1759)	48.1(214)	48.0(214)	0.2
6700(1964)	52.9(235)	52.9(235)	0.0
7200(2110)	57.0(254)	56.5(251)	0.9
8100(2374)	64.0(285)	62.9(280)	1.7

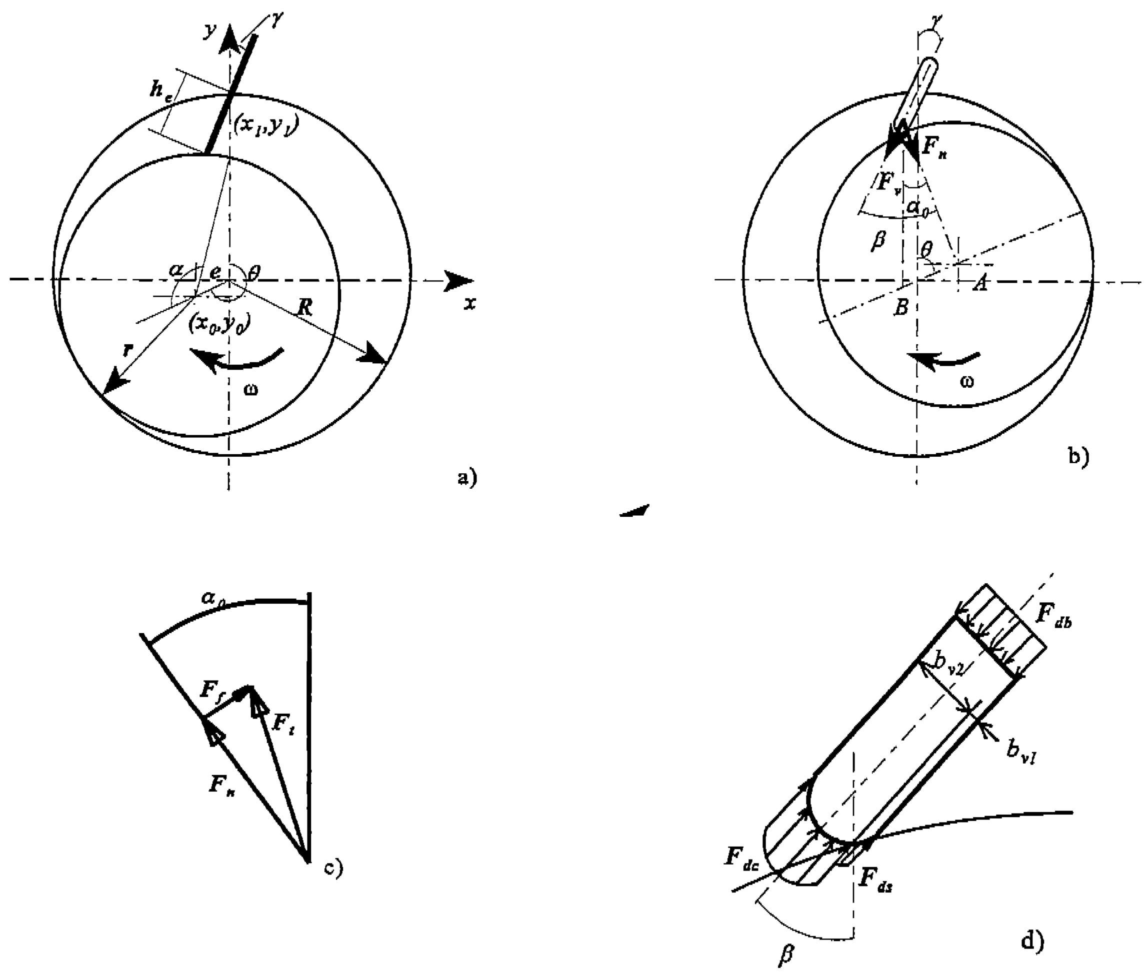

Figure 1 a) Geometric Relations in the Rotary Compressor; b) Force Relations between Vane and Roller; c) Vane Tip Force Composition; d) Gas Forces Acted in Vane Moving Direction.

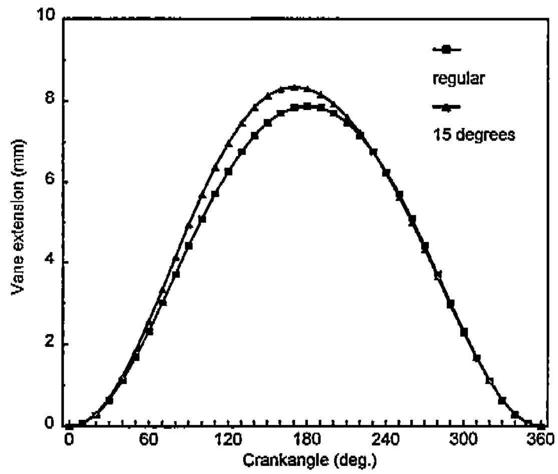

Figure 2 Vane extension vs. crankangle

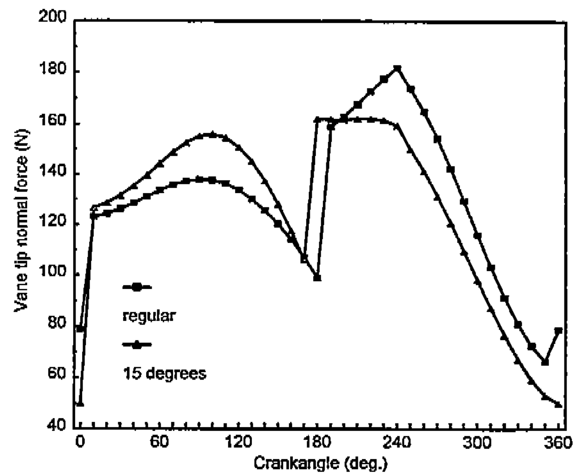

Figure 3 Vane tip normal force vs. crankangle

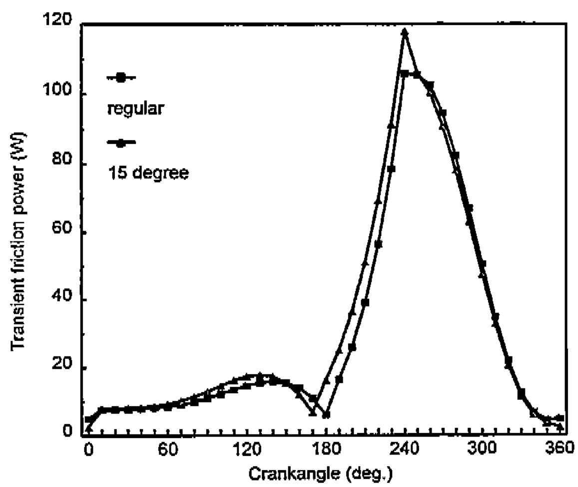

Figure 4 Transient friction power vs. crankangle

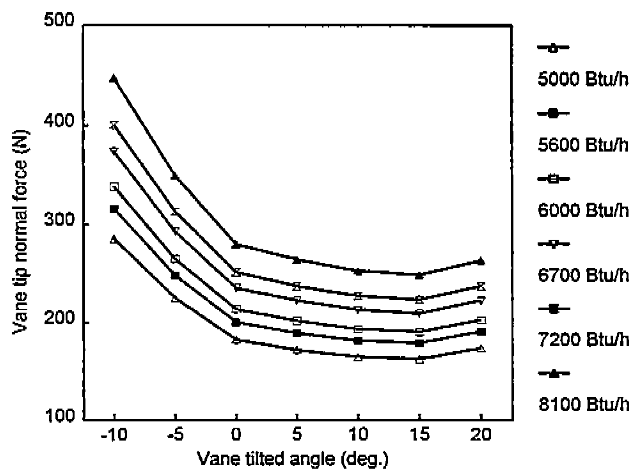

Figure 5 Vane tip normal force vs. vane tilted angle

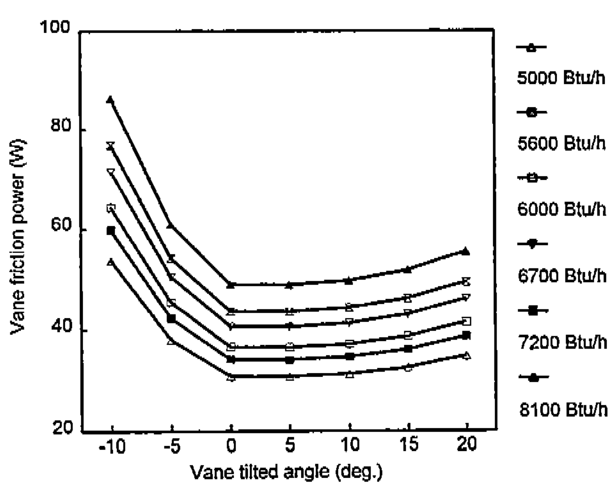

Figure 6 Vane friction power vs. vane tilted angle